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NONCIRCULAR BAR AND TUBING

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TORSION TESTS OF 24S-T ALUMINUM-ALLOY NONCIRCULAR BAR AND TUBING

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SUMMARY

Tests of 24S-T aluminum alloy have been made to determine the yield and ultimate strengths in torsion of noncircular bar and tubing. An approximate basis for predicting these torsional strength characteristics has been indicated. The results show that the torsional stiffness and maximum shearing stresses within the elastic range may be computed quite closely by means of existing formulas based on mathematical analysis and the membrane analogy.

INTRODUCTION

Torque-resisting members of noncircular section are frequently used in aircraft construction although there are apparently few experimental data on which to base the design. The formulas that have been developed for computing torsional stiffness and shearing stresses within the elastic range are based upon mathematical analysis or the membrane analogy and have had only a limited experimental verification. Little appears to be known about first yielding and the ultimate torsional strengths of noncircular sections of ductile materials. This lack of information seems rather significant in view of the numerous tests that have been made to determine the torsional strength of round tubing and the emphasis placed on such data in design. (See fig. 5-7 of reference 1.)

In long members of open section that have little torsional stiffness, computed behavior within the elastic range is probably a more important factor in design in most cases than yield or ultimate torsional strengths. In short lengths of more compact solid sections, however, such as may be used for fittings or brackets, yield and ultimate strengths in torsion may be more significant.

Requests for information on this phase of the torsion problem have led to an experimental investigation of the strength characteristics of a number of different non-circular sections. This report presents the results obtained from tests of rectangular bar and sections of square, rectangular, and streamline tubing.

The objects of this investigation were (1) to determine the yield and ultimate strengths in torsion of several noncircular sections of 24S-T aluminum-alloy bar and tubing and (2) to compare the measured torsional stiffness and shearing stresses in the elastic range with values computed by existing formulas for noncircular sections.

SPECIMENS

Torsion tests were made of the following 24S-T aluminum-alloy sections having nominal dimensions as given: Bar, $3/8$ by $1\frac{1}{2}$ inches; bar, $3/4$ by $1\frac{1}{2}$ inches; bar, $3/4$ by $3/4$ inch; square tubing, $1\frac{1}{4}$ by $1\frac{1}{4}$ by $1/8$ inch; rectangular tubing (T-568), $1\frac{1}{2}$ by $5/8$ by 0.065 inch; streamline tubing (T-158), $3\frac{1}{8}$ by $1\frac{3}{16}$ by 0.065 inch. A supplementary test was also made of a round specimen 0.739 -inch in diameter turned from the $3/4$ - by $3/4$ -inch square bar.

Table I summarizes the mechanical properties of the material used. All tensile values are considerably higher than specified minimum values for 24S-T bar and tubing. (See table 21 of reference 2.) Although the tensile yield strengths range from about 4000 to 10,000 pounds per square inch higher than the compressive yield strengths, the compressive strengths, with one exception, are also higher than the specified tensile-yield values.

Estimated values of shearing yield and ultimate strengths are included in table I because of their significance in the present investigation. Shearing yield strengths were assumed to be equal to one-half the tensile yield strengths, a ratio based upon the results of tension and torsion-shear tests of 24S-T round tubing having tensile properties comparable to those given in table I. The factor of 0.65 applied to the tensile strengths to obtain ultimate shear strengths is also based on the results of previous tests.

PROCEDURE

The torsion tests were made in an Amsler machine having ranges of capacity of 240, 400, 800, and 1200 foot-pounds. The specimens of rectangular bar were all 24 inches long and provided a clear length of 16 inches between grips. Two lengths of each tube section were tested: one 20 inches long, providing a clear length of 12 inches; the other 44 inches long, providing a clear length of 36 inches. All specimens were held in the torsion machine by the flat or V-grips provided. The ends of the tubes were reinforced with zinc plugs about 4 inches long that were precast in molds of the tube sections. In the case of the streamline tubes, steel plates, machined on one side to fit the contour of the tubing, were inserted between the specimens and the flat grips of the torsion machine as shown in figure 1.

Diagonal tensile and compressive strains were measured by Huggenberger Tensometers on 1/2-inch gage lengths, making an angle of 45° with the axis of twist. The locations of these gage lines are shown in figures 2 to 7. On all but the streamline tube, Tensometers were mounted on parallel gage lines on opposite sides of the specimens; one instrument showed tension, the other compressive strain. In most cases, both types of strain were measured on each side at the point of assumed maximum stress.

The stresses corresponding to the measured diagonal tensile and compressive strains were obtained by the relationship

$$\sigma_x = \frac{E}{1 - \mu^2} (\epsilon_x + \mu\epsilon_y)$$

where

σ_x diagonal tensile or compressive stress, pounds per square inch

E modulus of elasticity in tension or compression
assumed to be 10,500,000 pounds per square inch

μ Poisson's ratio, assumed to be 1/3

ϵ_x tensile or compressive strain on one diagonal gage line, inch per inch

and

ϵ_y strain on other diagonal gage line, inch per inch

Since the measured strains in the two directions were approximately equal but opposite in sign, a state of essentially pure shear was shown to exist and the shear stresses on sections normal to the axis of twist could be assumed equal in magnitude to the diagonal tensions and compressions.

Angles of twist were measured by Amsler troptometers, sensitive to about 0.1° . Gage lengths of 10 inches were used for the rectangular bars and gage lengths of 8 and 32 inches were used for the tubing.

RESULTS AND DISCUSSION

Figures 2 to 7 summarize the principal results of the twist and shear-stress measurements made on each section. Comparisons between measured and computed behavior within the elastic range are indicated in these figures as well as in table II. Since the torque-twist relationships found for the two lengths of each tube section tested were essentially the same, data for only the larger specimen are shown.

With the exception of the twist found for the $3/8$ - by $1\frac{1}{2}$ -inch rectangular bar, the observed behavior of the rectangular sections shown in figures 2 to 4 was almost the same as that computed. Angles of twist and maximum shearing stresses were computed by the following formulas (reference 3), based on the solution of Saint Venant:

For twist,

$$\theta = \frac{T}{\beta bc^3 G} \quad (1)$$

where

θ twist, radians per inch

T torque, pound-inches

b long side, inches

c short side, inches

G modulus of elasticity in shear, assumed to be 3,900,000 pounds per square inch

and

β factor depending upon ratio b/c

For the cases considered here, $\beta = 0.141$ for $b/c = 1$, $\beta = 0.229$ for $b/c = 2$, and $\beta = 0.281$ for $b/c = 4$.

For maximum shear stress at the center of the long side,

$$\tau_{\max} = \frac{T}{\alpha bc} \quad (2)$$

where $\tau_{\max}$ is the maximum shear stress in pounds per square inch and α is a factor depending upon the ratio b/c . For the cases considered here, $\alpha = 0.208$ for $b/c = 1$, $\alpha = 0.246$ for $b/c = 2$, and $\alpha = 0.282$ for $b/c = 4$.

It is evident from a consideration of the membrane analogy that the distribution of shear stress across the wide face of a narrow rectangular section should be uniform over the greater part of the width. The measured diagonal tensile and compressive stresses tabulated in figure 2, which give a measure of the shearing stresses developed on sections normal to the axis of twist, show that such a behavior was approached quite closely in the 3/8- by 1 1/2-inch bar. Although approximately the same values of maximum stress were found on the 3/4- by 1 1/2-inch bar, as shown in figure 3, the decrease in stresses toward the edges was more pronounced in the thicker section. The measurement of strains on a 1/2-inch gage length, of course, limited the degree to which the stress distribution could be determined, particularly across the shorter sides. Such values as were observed indicated the maximum stresses on the short sides to be about 70 percent of those on the long sides.

The agreement between measured and computed rotations for the tubing was not quite so satisfactory as for the rectangular sections although the maximum differences, as indicated in table II, did not exceed about 7 percent. Computed twists were determined by the following formulas (reference 3), based on the membrane analogy:

$$\theta = \frac{Tp}{4A Gt} \quad (3)$$

where

p mean perimeter of cross section, inches

A area enclosed by mean perimeter, square inches

and

t wall thickness, inches

The shearing stresses in thin-wall tubes are generally assumed to be uniformly distributed over the cross-sectional area and are given by the following formula, also based on the membrane analogy:

$$\tau = \frac{T}{2At} \quad (4)$$

where τ is the mean shear stress in pounds per square inch and the other terms are defined as for equation (3).

It will be noted from figures 5 and 6 that the diagonal tensile and compressive stresses measured in the square and the rectangular tubes were not altogether consistent with the distribution of shear stress assumed although, as indicated by a comparison of measured and computed torque-twist curves, the influence of this variation upon over-all behavior was not significant. The measured stresses at the edges were from about 5 to 9 percent less than found at the center of the sides and those measured at the center of the short sides of the rectangular tube were about 18 percent less than found at the corresponding location in the long sides. Corner effects, which are neglected in equation (4), may conceivably account for variations of the magnitude observed. The maximum difference in measured stress found for 14 determinations made over the surface of the streamline tube was 1200 pounds per square inch or about 8 percent of the highest value.

Probably of more significance is the fact that the maximum shearing stresses indicated were considerably higher than the mean shear stress computed by equation (4). The differences in the case of the square and rectangular tubes were about 25 percent and that in the case of the streamline tube about 10 percent. Previous tests have indicated that an appreciable difference may exist between the mean and the surface stresses in noncircular tubes (reference 4). Theoretical investigations (reference 5) have since shown that these differences in stress may be computed by the following modified form of equation (4):

$$\tau = \frac{T}{2At} \left[1 \pm \frac{t}{2} \left(\frac{p}{A} - \frac{1}{R} \right) \right] \quad (5)$$

where τ is the shear stress at the inner or outer surface in pounds per square inch, and R is the radius of curvature of the mean perimeter at the point considered. Figures 5 to 7 show a fairly good agreement between the measured stresses and those computed for the outside surfaces by equation (5).

Although considerable emphasis is placed upon first yielding or permanent set in the design of aircraft structural members, no generally accepted basis for evaluating yield characteristics has ever been established. Table III summarizes the torques corresponding to what appears to be first yielding, as indicated by the torque-twist curves in figures 2 to 7, and gives the corresponding computed maximum shearing stresses. The computed maximum shearing stresses are all less than the estimated shearing yield strengths given in table I and in most cases are within the range in which first yielding in shear would be expected from a consideration of the elastic properties of the materials. It is of interest to note that first yielding in the square tube was not apparent as early as in the rectangular tube, despite the fact that the stress concentrations at the sharp corners in the square tube were presumably much higher. According to equation (5), the outside stresses in the flat sides of the rectangular tubes were higher than at the inside at the corners. Likewise, the outside stresses in the flattest portion of the streamline tubes were computed to be higher than those inside at the sections of sharper curvature.

The moduli of failure in table III are given as a measure of the ultimate torsional strength of the sections tested. These values of stress were obtained by substituting ultimate torques in the formulas for stress previously referred to and which, of course, are strictly applicable only within the elastic range. The stress given for the solid round bar (included as an auxiliary test for comparative purposes) is the extreme fiber stress, computed by the ordinary torsion formula for circular sections.

The moduli of failure computed for the solid sections ranged from about 1.2 to 2.4 times the estimated shear strengths of the material given in table I. It will be appreciated from the large deformations shown at failure in figure 8 that this apparent difference in strengths may

be attributed mainly to the fact that the distribution of stress actually obtained at failure was quite different from that assumed, not only from a consideration of shear but also so far as secondary stress effects resulting from large angles of twist were concerned.

It has been concluded from previous torsion tests that the shear stress distribution produced in round bars of ductile material must be very nearly uniform at the time of failure. It seems reasonable to believe that a similar stress distribution was also approached in these tests of rectangular bars. Table IV shows how closely the observed ultimate torques may be computed using the estimated values of shear strength given in table I and assuming a uniform distribution. The fact that the actual torques obtained for the sections which failed by shear ranged from 82 to 88 percent of the computed values indicates that uniform stress conditions were not quite realized. The fairly constant relationship found, however, is believed to provide a more practical basis for predicting ultimate torsional strengths for rectangular bars than may be obtained from a consideration of the range of moduli of failure given in table III. Some difference in ratios of actual to predicted ultimate torques would be expected in materials having appreciably more or less ductility than the 24S-T alloy considered, although until more data are available a ratio of 0.85 seems reasonable.

The procedure followed in computing ultimate torques for rectangular sections on the basis of a uniform stress distribution was simply to divide the cross section into four triangular areas by diagonals across the corners and to compute the total shear force developed on each. The resisting moment of the section was assumed to be equal to the sum of the moments of these four shearing forces, acting at the centers of gravity of the corresponding triangular areas.

The moduli of failure shown in table III for the tubing specimens were computed by equation (4), which assumes a uniform distribution of shear stress. None of these moduli of failure values equaled the estimated shear strengths of the material given in table I, although the percentage of this value developed in the square tubes (83 percent) was about the same as found in the case of the solid sections.

Figure 9 shows the type of failure obtained in the tubing specimens. The rectangular tubes failed by buckling

of the tube walls at average stresses equal approximately to the estimated yield strength of the material in shear, which corresponds closely to the strength computed from a consideration of the shear-buckling resistance of the long sides, treated as flat-sheet panels with simply supported edges. (See table 17 of reference 6.) The moduli of failure for the streamline tubes, which also failed by buckling, were in fair agreement with the torsional strength computed for a circumscribed circular tube having the same length and thickness, a computation procedure suggested by the results of tests of cylinders of elliptic section (reference 7). Additional tests are required, of course, to establish the general validity of this method for streamline tubes.

It will be noted from table III that the lengths of tubing tested had no significant influence upon the torsional strengths obtained.

CONCLUSIONS

The following conclusions are based upon the results of the torsion tests of 24S-T aluminum-alloy bar and tubing described in the present report:

1. Within the elastic range, torsional stiffness and maximum shearing stresses may be predicted quite closely in solid rectangular sections and in square, rectangular, and streamline tubing by means of existing formulas based upon mathematical analysis and the membrane analogy.
2. In sections of the type investigated, the limit of the elastic range or the point of first yielding in torsion may be expected at shearing stresses somewhat less than the shearing yield strength of the material based upon a 0.2-percent-offset criterion. As far as could be determined from the over-all torque-twist characteristics observed in these tests, yielding occurred no earlier in the square tubes with sharp corners than in the rectangular tubes with round corners.
3. The stresses developed in the outside surface of noncircular tubes may be appreciably greater (about 25 percent in these tests) than the mean value which is generally computed by the equation for the shearing stresses in thin-wall tubes. The stresses measured in these tests were in good agreement with those computed by the less familiar equation for the shear stress at the inner or outer surface.

4. The values of modulus of failure computed for the solid bars by substituting ultimate torques in the stress formulas for elastic action were found to range from 1.2 to 2.4 times the estimated ultimate shear strengths of the material. An approximate ultimate-strength criterion for 24S-T bars can apparently be based on the fact that the torques producing failure in all the solid sections averaged about 85 percent of those computed, assuming a uniform distribution of shear stress at failure equal to the estimated shear strength of the material.

5. The values of modulus of failure computed by the equation for the mean shear stress for the square tubes, which were the only specimens in this group to fail by fracture, were also about 85 percent of the estimated shear strengths.

6. The values of modulus of failure computed for the rectangular tubes, which failed by buckling, were in reasonably good agreement with the computed shear-buckling resistance of the sides, considered as flat panels with simply supported edges.

7. The values of modulus of failure computed for the streamline tubes, which also failed by buckling, were approximately the same as computed for a circumscribed circular tube having the same length and thickness. This computation procedure was suggested from observations made regarding the torsional strength of cylinders of elliptic section, but needs further proof of its general applicability to streamline tubes.

Aluminum Research Laboratories,
Aluminum Company of America,
New Kensington, Pa., November 2, 1942.

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TABLE I

MECHANICAL PROPERTIES OF 24S-T ALUMINUM-ALLOY BAR AND TUBING

Section (nominal dimensions)	Tension			Compression		Shear ^a	
	Yield strength (offset = 0.2 percent) (lb/sq in.)	Ultimate strength (lb/sq in.)	Elongation in 2 in. (percent)	Yield strength (offset=0.2 percent) (lb/sq in.)	Ultimate strength (lb/sq in.)	Yield strength (lb/sq in.)	Ultimate strength (lb/sq in.)
Bar, 3/8 by 1½ in.	49,200	71,200	20.5	45,300	-----	24,600	46,300
Bar, 3/4 by 1½ in.	46,000	66,800	^b 21.1	39,800	-----	23,000	43,400
Bar, 3/4 by 3/4 in.	57,700	72,000	^b 16.7	48,300	-----	28,800	46,800
Square tubing, 1¼ by 1¼ by 1/8 in.	54,600	72,300	18.5	44,000	78,100	27,300	47,000
Rectangular tubing (T-568) 1½ by 5/8 by 0.065 in.	54,000	74,100	18.0	46,000	54,000	27,000	48,100
Streamline tubing (T-158) 3⅛ by 1⅜ by 0.065 in.	54,300	73,100	18.5	^c 46,500	42,500	27,200	47,500

^aEstimated.

Yield strength in shear = 0.5 × yield strength in tension.

Ultimate strength in shear = 0.65 × ultimate strength in tension.

^bElongation in 4D.^cYield strength value determined from test of specimen cut from tube, using single-thickness method.

All other compressive values for tubing are from tests of full sections, 5 in. long.

TABLE II

COMPARISON OF MEASURED AND COMPUTED TWISTS AND SHEARING STRESSES FOR 24S-T ALUMINUM-ALLOY BAR AND TUBING

Section (nominal dimensions)	Torque, ^a (lb-in.)	Twist (deg per in.)		$\frac{\text{Measured twist}}{\text{Computed twist}}$	Shear stress ^b (lb/sq in.)		$\frac{\text{Measured stress}}{\text{Computed stress}}$
		Measured	Computed		Measured	Computed	
Bar, $3/8$ by $1\frac{1}{2}$ in.	866	0.61	0.56	1.09	14,900	14,400	1.03
Bar, $3/4$ by $1\frac{1}{2}$ in.	3048	.31	.31	1.00	14,700	14,600	1.01
Bar, $3/4$ by $3/4$ in.	1218	.41	.41	1.00	13,700	14,000	.98
Square tubing, $1\frac{1}{4}$ by $1\frac{1}{4}$ by $1/8$ in.	4877	.39	.42	.93	20,500	19,500	1.05
Rectangular tubing (T-568), $1\frac{1}{2}$ by $5/8$ by 0.065 in.	1475	.49	.49	1.00	17,400	16,200	1.07
Streamline tubing (T-158), $3\frac{1}{8}$ by $1\frac{3}{16}$ by 0.065 in.	4788	.27	.29	.93	15,200	15,200	1.00

^aMaximum torques for stress measurements.^bLocations considered are the same as those for which torque-stress curves are shown in figs. 2 to 7.
Computed stresses for tubing are for outside surfaces.

TABLE III

TORSIONAL STRENGTH OF 24S-T ALUMINUM-ALLOY BAR AND TUBING

Section (nominal dimensions)	Length between grips (in.)	Estimated first yielding ^a (lb-in.)	Corresponding computed shear stress (lb/sq in.)	Ultimate torque (lb-in.)	Corresponding modulus of failure ^b (lb/sq in.)	Type of failure
Bar, 3/8 by 1 1/2 in.	16	1400	23,200	6,720	111,700	Fracture
Bar, 3/4 by 1 1/2 in.	16	4000	19,100	^c 14,400	69,000	Large twist
Bar, 3/4 by 3/4 in.	16	2000	23,000	5,688	65,400	Fracture
Bar, 0.739-in. diameter round from 3/4 by 3/4 in.	16	1390	17,500	4,390	55,400	Fracture
Square tubing, 1 1/4 by 1 1/4 by 1/8 in.	12 36	----- 5000	----- 20,000	11,950 11,760	39,200 38,600	Fracture Fracture
Rectangular tubing (T-568), 1 1/2 by 5/8 by 0.065 in.	12 36	----- 1700	----- 18,700	3,000 2,940	28,400 27,800	Buckling Buckling
Streamline tubing (T-158), 3 1/8 by 1 3/16 by 0.065 in.	12 36	----- 4000	----- 12,700	7,260 7,285	21,200 21,300	Buckling Buckling

^aEstimated from torque-twist curves in figs. 2 to 7. Value for round bar corresponds to proportional limit indicated on shear stress-strain curve not shown.

^bValues obtained by substituting ultimate torques in equations (2) and (4).

^cCapacity of torsion machine; not sufficient to produce fracture of specimen.

TABLE IV

COMPARISON OF OBSERVED AND COMPUTED ULTIMATE
TORQUES FOR 24S-T ALUMINUM-ALLOY BAR

Torque (lb-in.) Section	Observed	Computed ^a	<u>Observed</u> <u>Computed</u>
3/8 by 1½ in.	6,720	8,190	0.82
3/4 by 1½ in.	^b 14,400	18,400	----
3/4 by 3/4 in.	5,688	6,550	.87
0.739-in. diam.	4,390	4,960	.88

^aBased on estimated ultimate shear strengths given in table I, assuming a uniform stress distribution.

^bCapacity of torsion machine; not sufficient to produce failure.

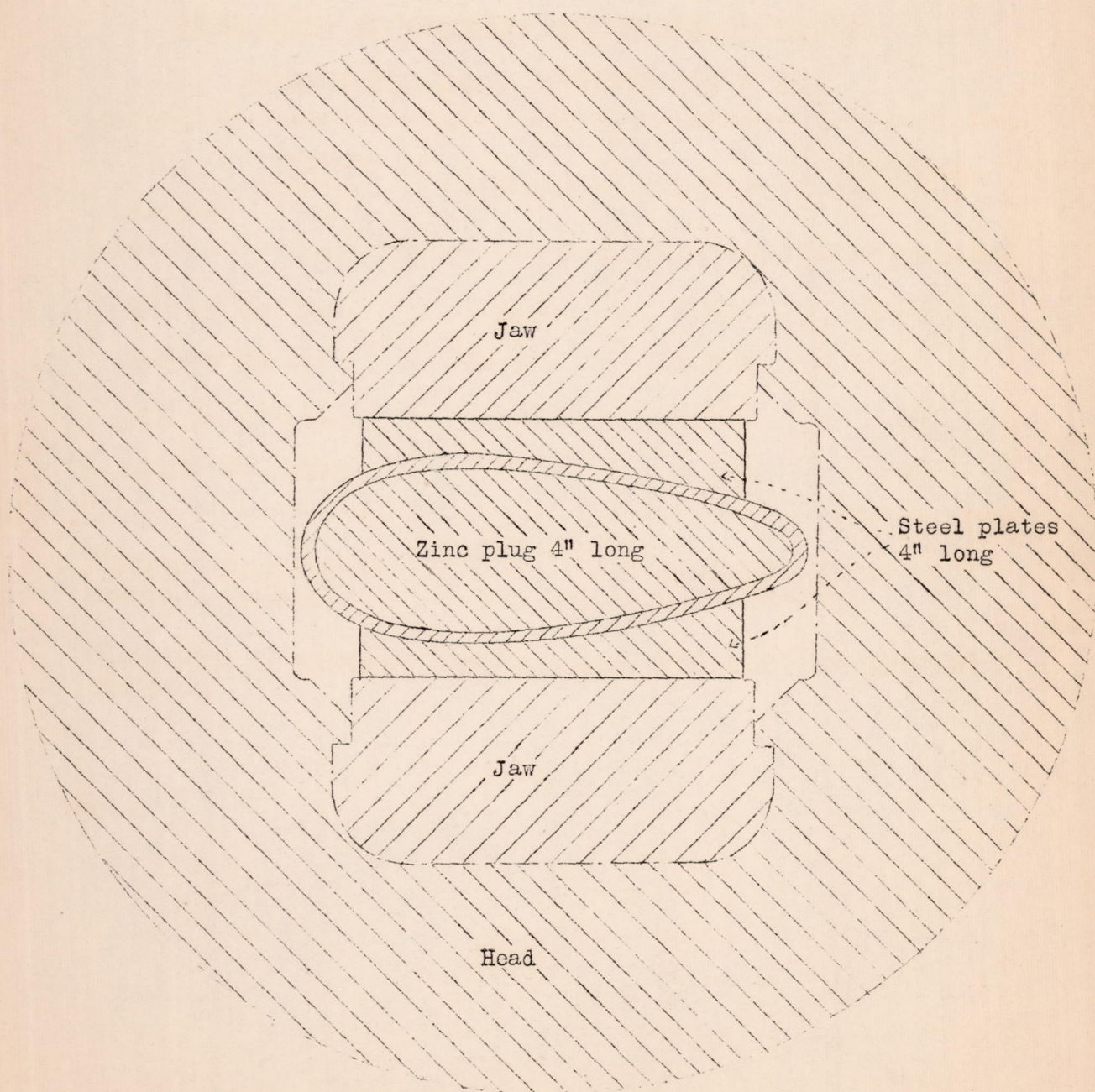


Figure 1.- Method of gripping streamline tube in torsion machine.

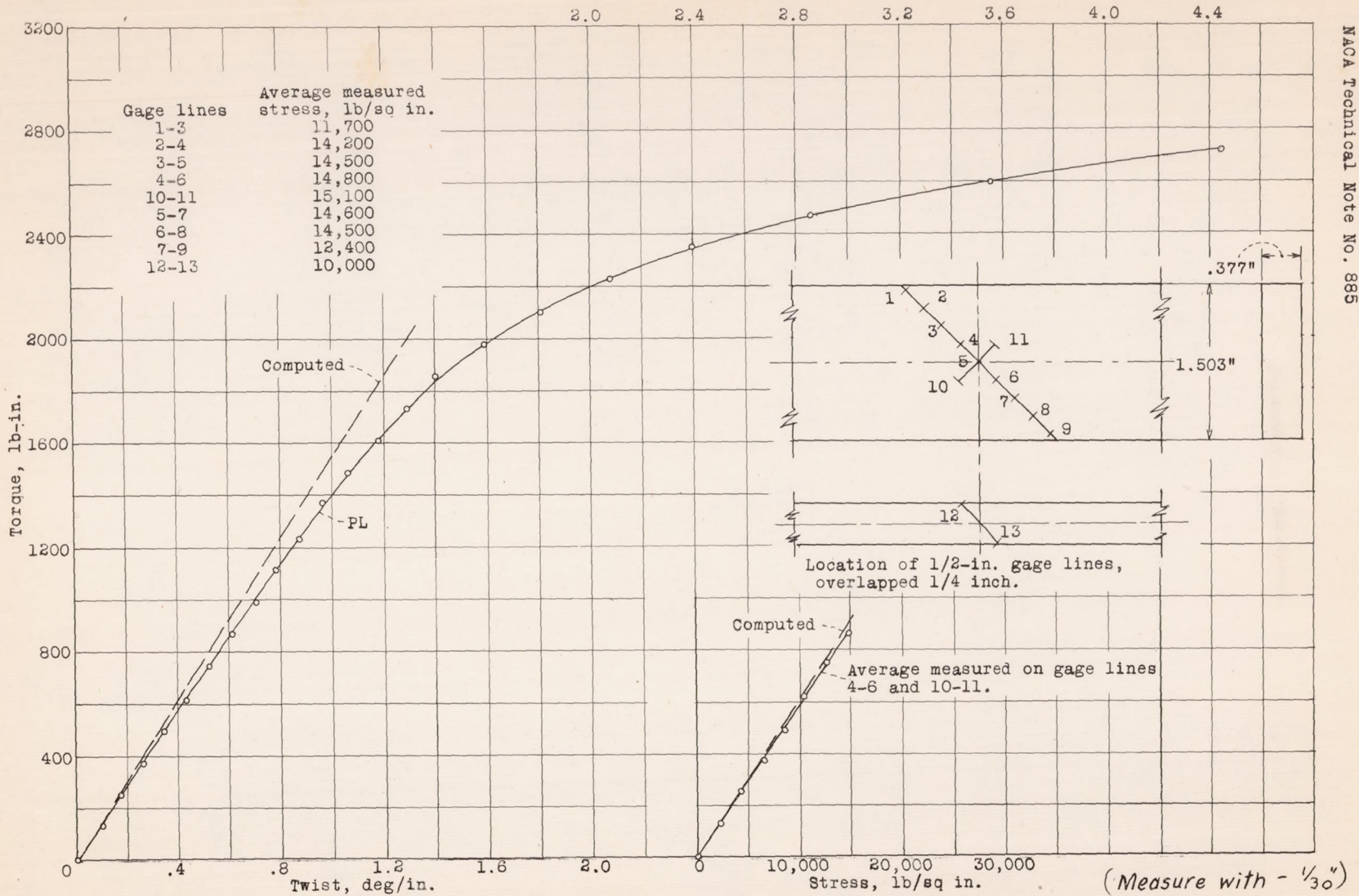


Figure 2.- Torsion-test data for 24S-T aluminum-alloy rectangular bar, 3/8 by 1-1/8 inches. The average measured stresses tabulated are the averages of diagonal tensile and compressive stresses measured on opposite sides of bar for a torque of 866 pound-inches; computed shear stress at center of long side, 14,400 pounds per square inch; gage length, 10 inches.

corresponding for twist measurements

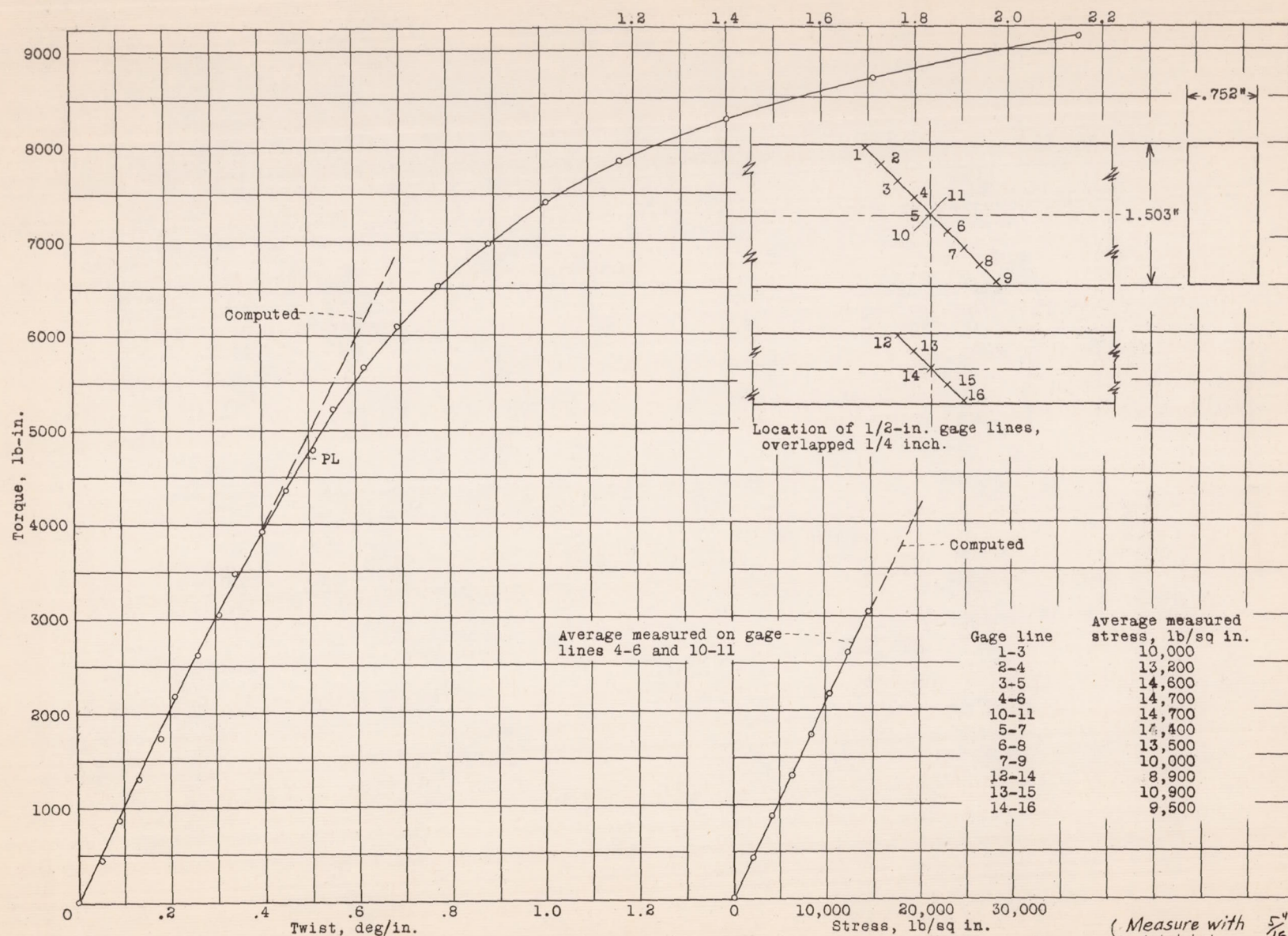


Figure 3.- Torsion-test data for 24S-T aluminum-alloy rectangular bar, 3/4 by 1-1/2 inches. The average measured stresses, the averages of diagonal tensile and compressive stresses measured on opposite sides of bar for a torque of 3058 pound-inches; a computed shear stress at center of long side, 14,600 pounds per square inch; gage length, 10 inches.

(Measure with $\frac{5}{16}$ tabulated are corresponding for twist measurements)

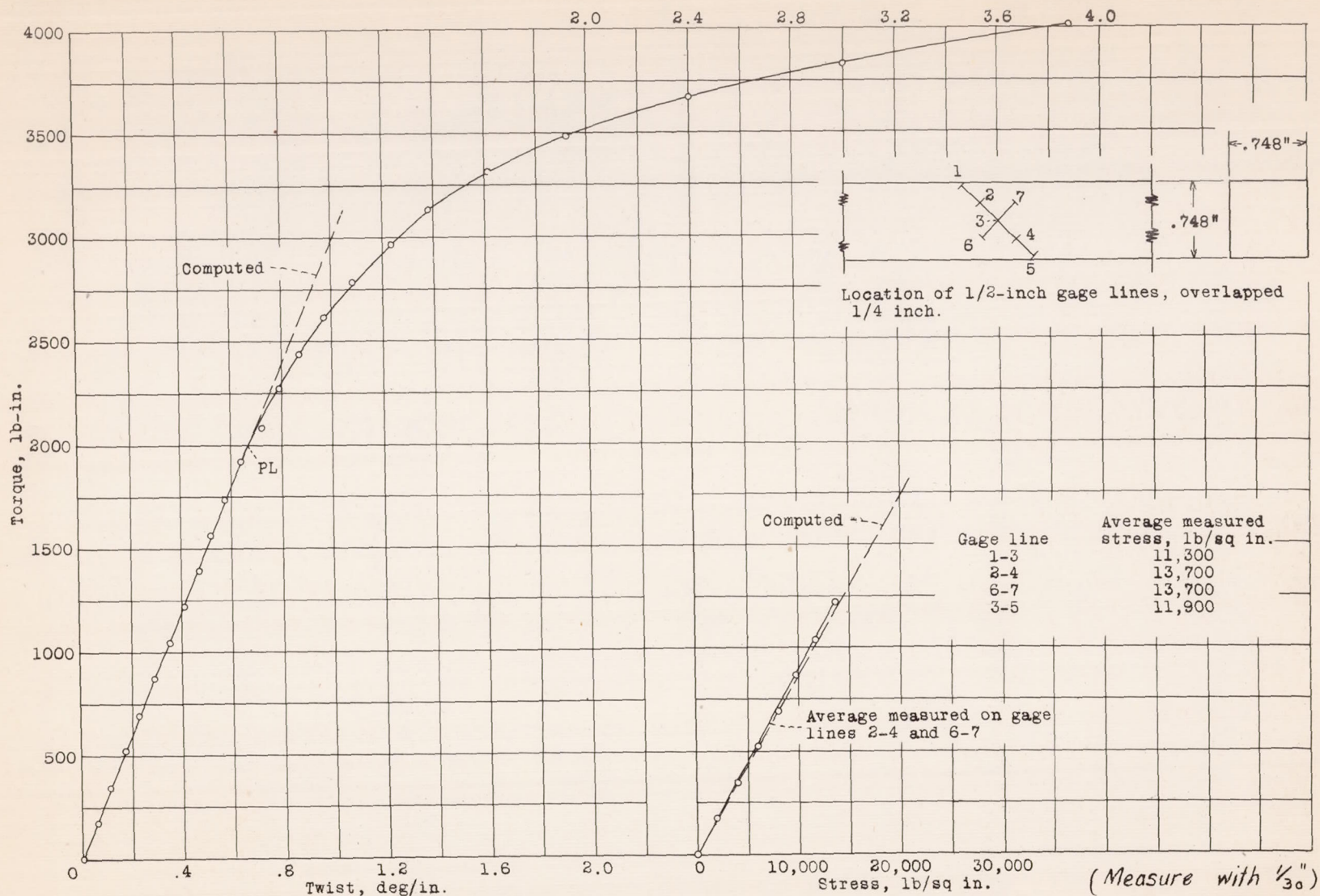


Figure 4.- Torsion-test data for 24S-T aluminum-alloy square bar, 3/4 by 3/4 inch. The average measured stresses tabulated are averages of diagonal tensile and compressive stresses measured on opposite sides of bar for a torque of 1218 pound-inches; computed shear stress at center of sides, 14,000 pounds per square inch; gage length, 10 inches.

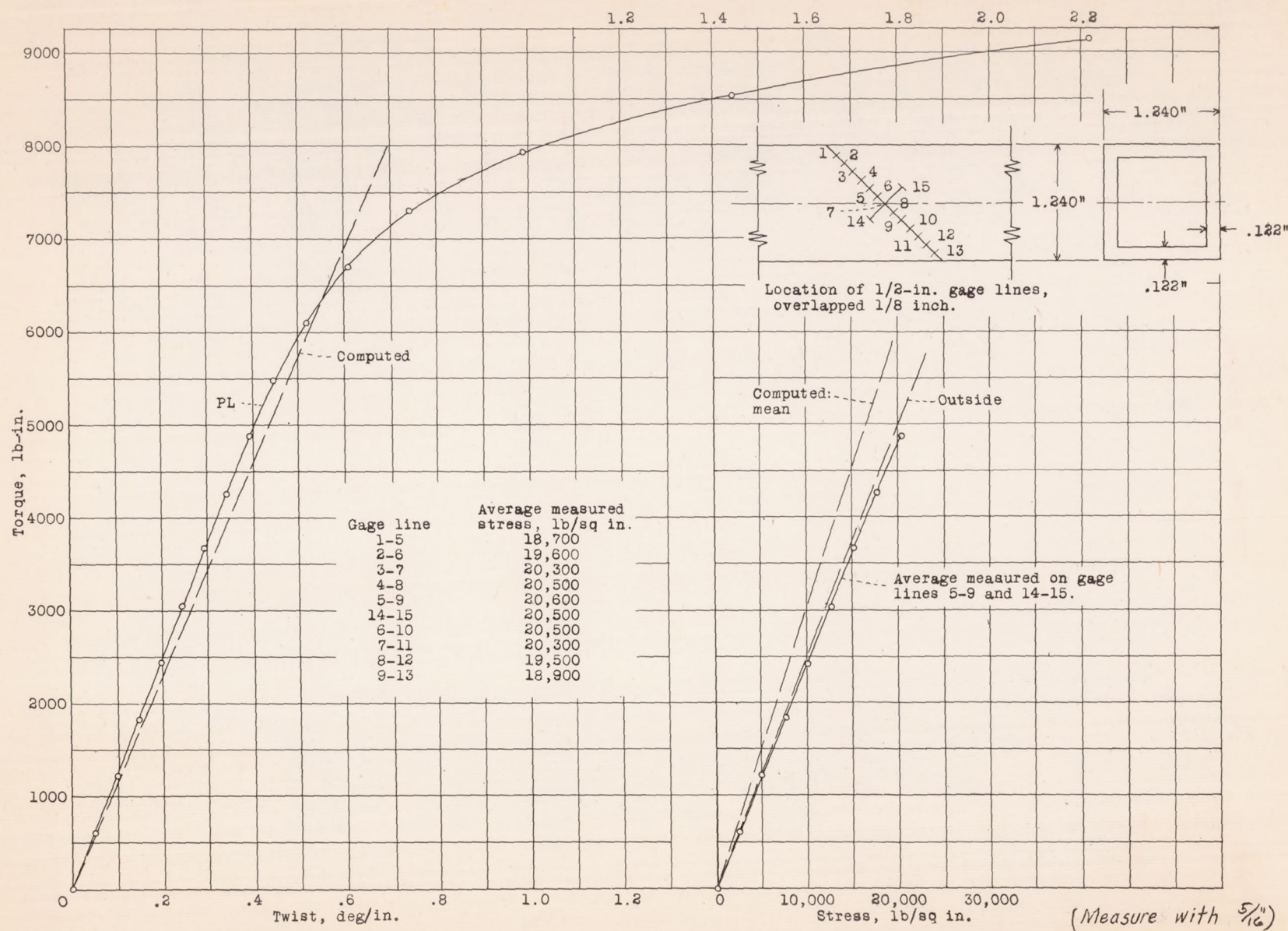


Figure 5.- Torsion-test data for 24S-T aluminum-alloy square tubing, 1-1/4 by 1-1/4 by 1/8 inch. The average measured stresses tabulated are averages of diagonal tensile and compressive stresses measured on opposite sides of tube for a torque of 4877 pound-inches; corresponding computed mean shear stress, 16,000 pounds per square inch; computed outside shear stress, 19,500 lb/sq in.; gage length, 32 inches. for twist measurements

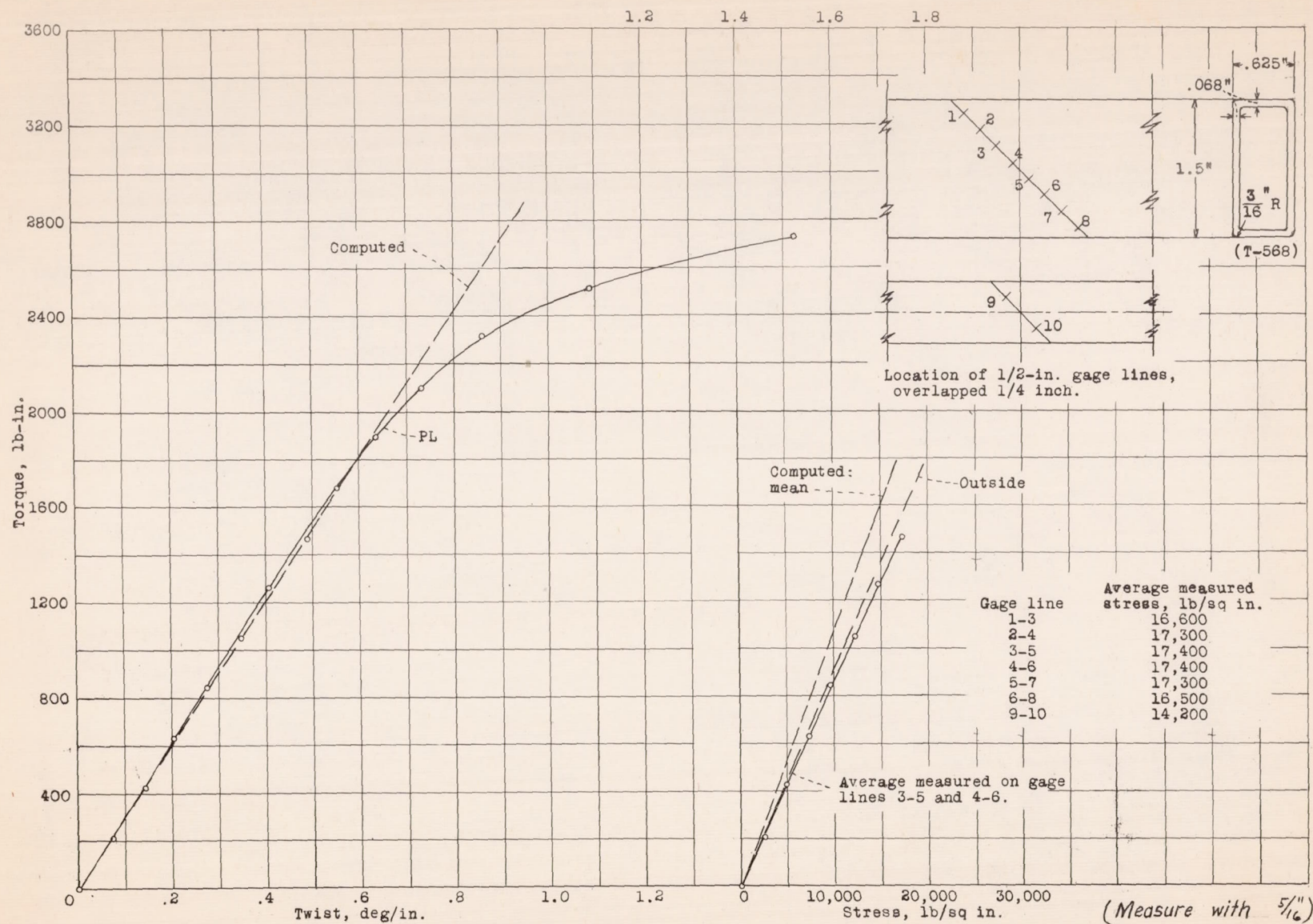


Fig. 6

Figure 6.- Torsion-test data for 24S-T aluminum-alloy rectangular tubing, 1-1/2 by 5/8 by 1/16 inch. The average measured stresses tabulated are the averages of diagonal tensile and compressive stresses measured on opposite sides of tube for a torque of 1475 pound-inches; a computed mean shear stress, 14,000 pounds per square inch; computed outside shear stress, 16,200 lb/sq in.; gage length, 32 in. corresponding

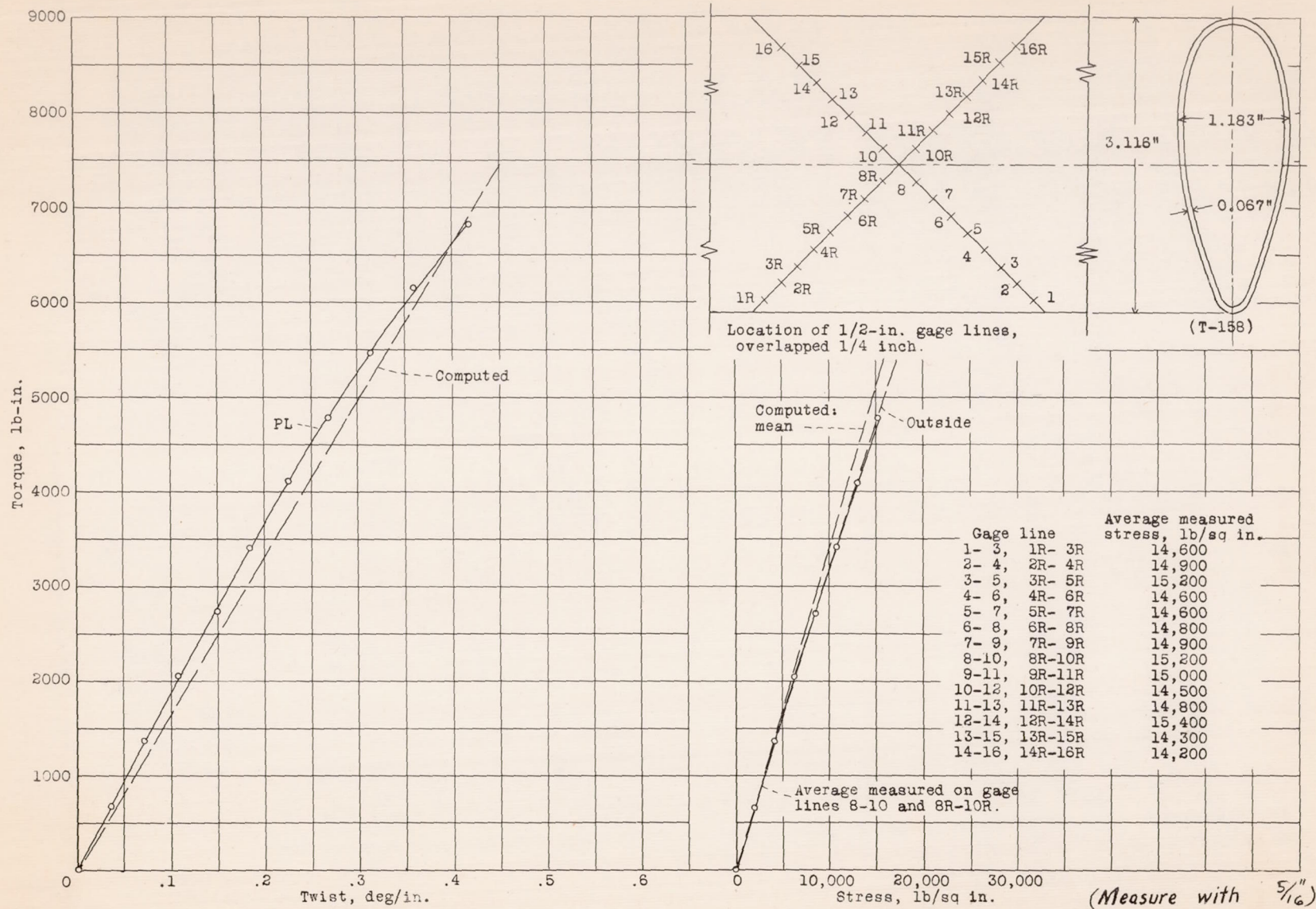
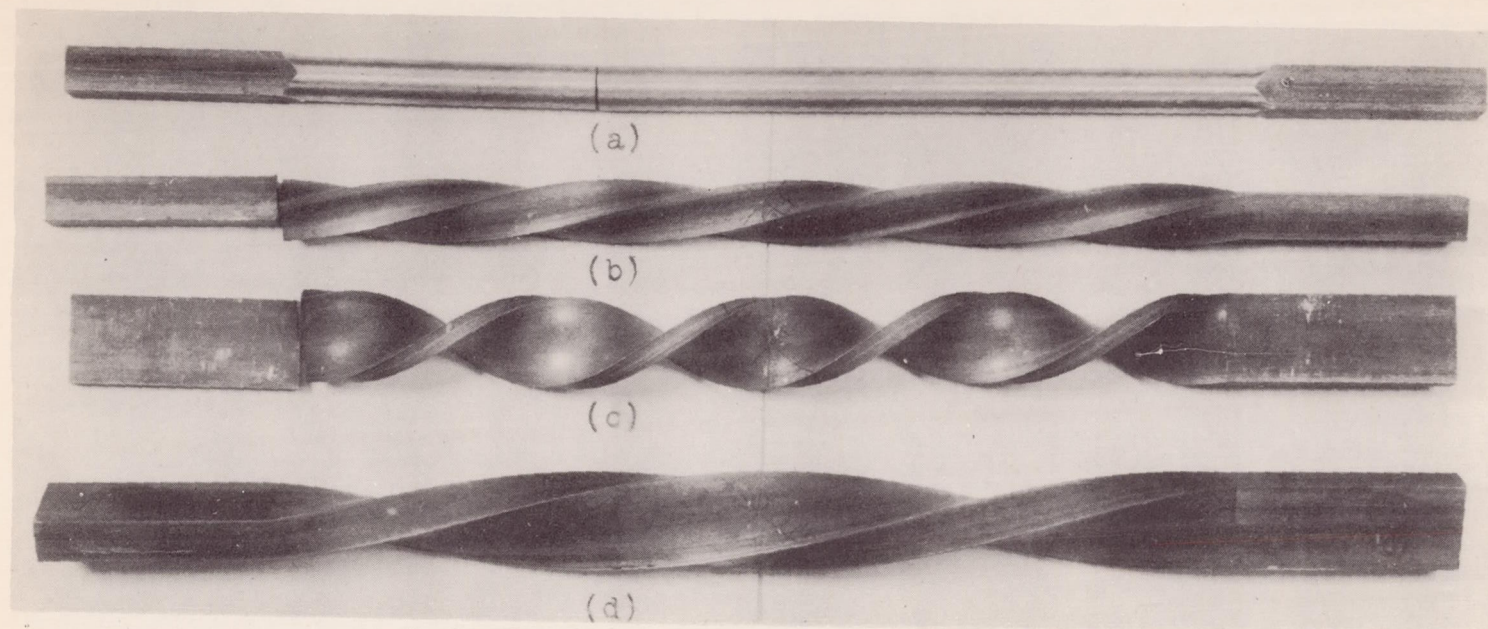


Figure 7.- Torsion-test data for 24S-T aluminum-alloy streamline tubing, 3-1/8 by 1-3/16 by 1/16 inch. The average measured stresses tabulated are the averages of diagonal tensile and compressive stresses measured on same side of tube for a torque of 4788 pound-inches; corresponding computed mean shear stress, 14,000 pounds per square inch; computed outside shear stress, 15,200 lb/sq in.; gage length, 32 inches. for twist measurements



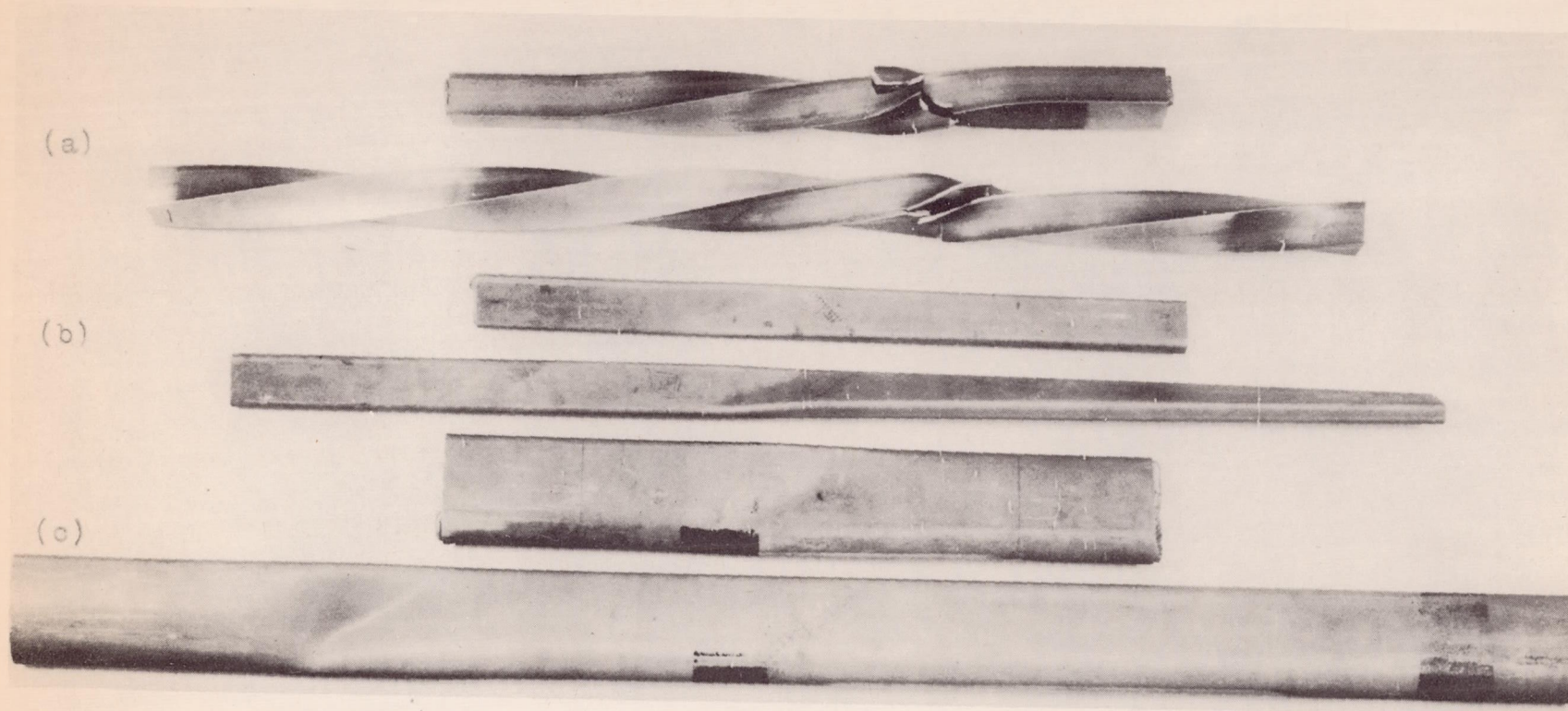
(a) 0.739-inch diameter.

(c) $\frac{3}{8}$ by $1\frac{1}{2}$ inches.

(b) $\frac{3}{4}$ by $\frac{3}{4}$ inch.

(d) $\frac{3}{4}$ by $1\frac{1}{2}$ inches.

Figure 8.- Specimens of 24S-T aluminum-alloy bar after torsion tests.



(a) $1\frac{1}{4}$ by $1\frac{1}{4}$ by $1/8$ inch. (b) $1\frac{1}{2}$ by $5/8$ by $1/16$ inch (c) $3\frac{1}{8}$ by $1\frac{3}{16}$ by $1/16$ inch

Figure 9.- Specimen~~s~~ of 24S-T aluminum-alloy tubing after torsion tests.